Design of an air distribution system for a multi-story office building

A

THESIS SUBMITTED IN PARTIAL FULFILLMENT OF

THE REQUIREMENTS FOR THE DEGREE OF

Master of Technology

In

Mechanical Engineering

(Specialization: Thermal Engineering)

By

CHUNESHWAR LAL VERMA

(ROLL NO 212ME3324)

DEPARTMENT OF MECHANICAL ENGINEERING

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CERTIFICATE

*This is to certify that the thesis entitled, "**Design of an air distribution system for a multi-story office building**" submitted by **Mr. Chuneshwar Lal Verma** in partial fulfillment of the requirements for the award of Master of Technology in Mechanical Engineering with Thermal Engineering specialization during session 2013-2014 in the Department of Mechanical Engineering, National Institute of Technology, Rourkela.*

It is an authentic work carried out by him under my supervision and guidance. To the best of my knowledge, the matter embodied in this thesis has not been submitted to any other University/Institute for the award of any Degree or Diploma.

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Place: Rourkela

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CHUNESHWAR LAL VERMA

ABSTRACT

Earlier the use of air conditioning for comfort purpose was considered to be expensive, but now-a-day, it has been a necessity for all human beings. Window air conditioners, split air conditioners are used in small buildings, offices etc. But, when the cooling load required is very high such as big buildings, multiplex, multi-story buildings, hospitals etc. centralized unit (central air conditioners) used. The central AC's systems are installed away from building called central plant where water or air is to be cooled. This cooled air not directly supplied to the building rooms. When the cooled air cannot be supplied directly from the air conditioning equipment to the space to be cooled, then the ducts are provided. The duct systems carry the cooled air from the air conditioning equipment for the proper distribution to rooms and also carry the return air from the room back to the air conditioning equipment for recirculation. When ducts are not properly designed, then it will lead to problem such as frictional loss, higher installation cost, increased noise and power consumption, uneven cooling in the cooling space. For minimizing this problem, a proper design of duct is needed. Equal friction method is used to design the duct, which is simple method as compared with the other design methods. These work gives the combination of theoretical and software tool to provide a comparative analysis of the duct size. It also gives the comparison between rectangular duct and circular duct.

Keywords – Equal friction method, friction loss, duct sizing.

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CHAPTER – 1

INTRODUCTION

1.1. General

In the present day, as the population increases the need for comfortness also increases. The human being needs more comfortness because of inferior environment (like light, sound, machine which produce heat). Sound, light and heat affect human comfort a lot. They may adversely affect the human comfort positively or negatively. Researchers suggest that, human body is used to be comfortable at a temperature of 22^{o}C to 25^{o}C. When the temperature of room is lower or higher than this temperature, than the human body feels uncomfortable. This is because, the human body is structured in a way that, it should receive a certain amount of light, failure to which it can cause sunburns and other skin conditions.

There are many types of air conditioning system like window air conditioners, split air conditioners etc. but these AC's system are used in small room or office where cooling load required is low. When the cooling load required is very high like multiplex building, hospital etc. central AC's system are used. In central AC's system the cooled air is directly not distributed to the rooms. The cooled air from the air conditioning equipment must be properly distributed to rooms or spaces to be cold in order to provide comfort condition. When the cooled air cannot be supplied directly from the air conditioning equipment to the spaces to be cooled, then the ducts are installed. The duct systems convey the cold air from the air conditioning equipment to the proper air distribution point and also carry the return air from the room back to the air conditioning equipment for reconditioning and recirculation.

As the duct system for the proper distribution of cold air, costs nearly 20% to 30% of the total cost of the equipment required. Thus, it is necessary to design the air duct system in such a way that the capital cost of ducts and the cost of running the fans is lower.

1.2. Classification of ducts

The duct may be classified as follows:

1.2.1. Supply air duct – The duct which supplies the conditioned air from the air conditioning equipment to the space to be cooled is called supply air duct.

1.2.2. Return air duct – The duct which carries the reciprocating air from the conditioned space back to the air conditioning equipment is called return air duct.

1.2.3. Fresh air duct – The duct which carries the outside air is called fresh air duct.

1.2.4. Low pressure duct – When the static pressure in the duct is less than 50 mm of water gauge, the duct is said to be a low pressure duct.

1.2.5. Medium pressure duct - When the static pressure in the duct is up to 150 mm of water gauge, the duct is said to be a medium pressure duct.

1.2.6. High pressure duct - When the static pressure in the duct is from 150-250 mm of water gauge, the duct is said to be a high pressure duct.

1.2.7. Low velocity duct – When the velocity of air in the duct is up to 600 m/min, the duct is said to be a low velocity duct.

1.2.8. High velocity duct - When the velocity of air in the duct is more than 600 m/min, the duct is said to be a high velocity duct.

1.3. Duct Material

The ducts are usually made from galvanized iron sheet metal, aluminium sheet metal or black sheet. The most commonly used duct material in the air conditioning system is galvanized sheet metal, because the zinc coating of this metal prevents rusting avoids the cost of painting. The sheet thickness of galvanized iron duct varies from 0.55 mm to 1.6 mm. The aluminium is used because of its lighter weight and resistance to moisture. The black sheet metal is always painted unless they withstand high temperature [30].

Now a day, the use of non-metal ducts has increased. The resin bounded glasses are used because they are quite strong and easy to manufacture according to the desired shape and size. They are used in low velocity application less than 600 m/min and for a static pressure below 5 mm of water gauge. Sometimes cement asbestos duct also used for underground air distribution. The wooden duct may be used in places where moisture content in the air is not very large [30].

1.4. Duct Shape

Fig 1.1 shows different duct shapes [31] and they are discussed in the following subsections:

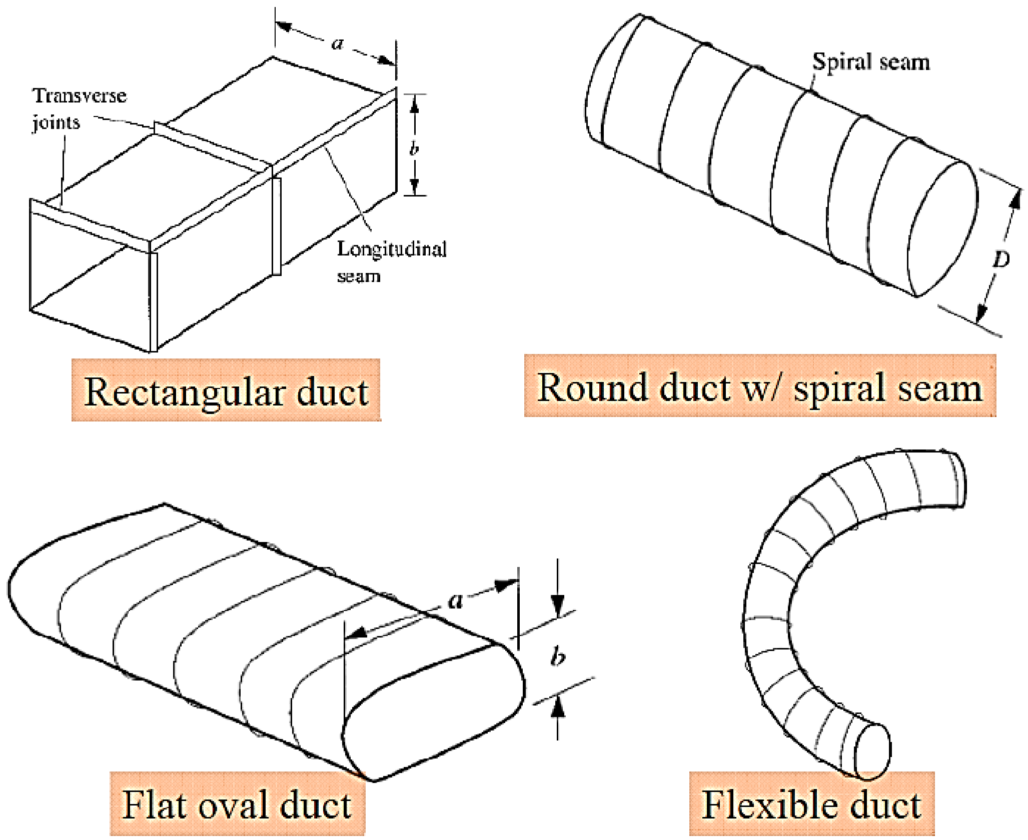

Fig.1.1. Various shapes of duct

1.4.1. Circular/round duct:

For a definite cross-sectional area and mean air velocity, a circular duct has less fluid resistance against air flow than other ducts. It also has better stiffness and strength. The longitude-seamed round ducts and spiral duct are used in commercial buildings. The main drawback of round ducts over the other duct is the more space required during installation.

1.4.2. Rectangular duct:

Rectangular duct takes less space as compared to the round duct. It can be easily fitted where space is less. Rectangular ducts are less stiff than circular ducts, and also easily fabricated. The air leakage in joint of rectangular ducts has a greater percentage than other ducts joint. The rectangular ducts are generally used in low-pressure systems.

1.4.3. Flat oval duct:

Flat oval ducts have a shape between round and rectangular cross-sectional shown in Fig. 1.1. Flat oval duct have the benefits of both the rectangular and the round duct with less large-scale air disorder and a lesser depth of space required. These ducts are easy to fit and also have lesser air leakage.

1.4.4. Flexible duct:

Flexible ducts are used to connect the key duct to the incurable (terminal) box. Their plasticity and ease of elimination allow separation and rearrangement of the incurable (terminal) devices. These ducts are made of numerous-ply polyester film reinforced by a helical steel wire core.

From an economical point of view, the circular ducts are preferable because the circular shape can carry more air in less space. This means that less duct material, less duct surface friction, and less insulation is required. Also, the circular ducts have less friction drop than the rectangular ducts.

1.5. Fan Coil Unit (FCU)

A fan coil unit (FCU) is a device consisting of a cooling or heating coil and fan. It is a part of the heating ventilation and air conditioning system used to circulate the cold water into the room. In FCU no need to ductwork and it is used to govern the temperature in the region where it is fitted. It is controlled by either physically or by a regulator.

Fan coil units (FCU) are normally used in places where economic installations are desired such as storage rooms, loading docks and corridors. In high-rise buildings, fan coils may be arranged, situated one above the another from floor to floor and all interrelated by the same tubing loop. FCUs are an admirable delivery apparatus for hydraulic chiller boiler systems in large housing and light profitable applications. In these applications the FCUs are mounted in bathroom ceilings and can be used to provide infinite comfort zones - with the facility to turn off vacant areas of the building to save energy.

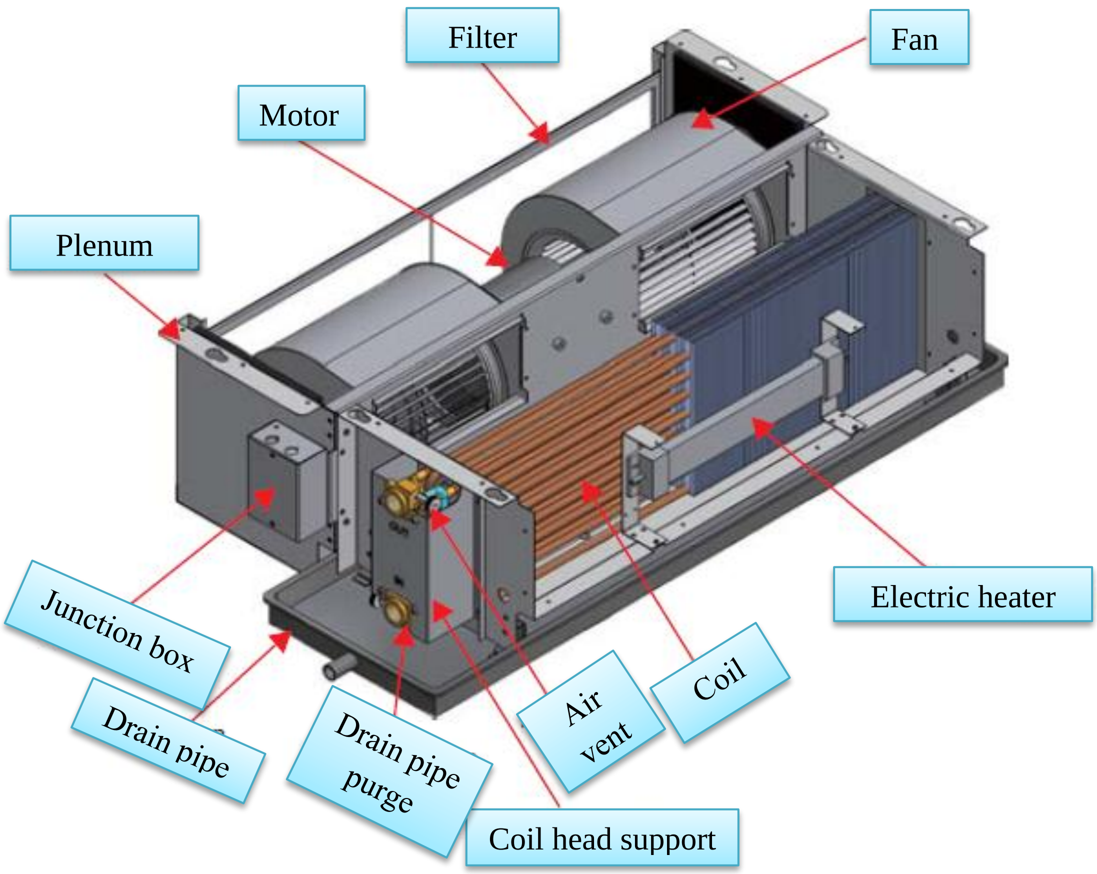

Fig.1.2. – Fan Coil Unit

1.5.1. Advantages of FCU

There are some advantages with the FCU which are given below:

- The system requires only piping installation which takes up less space than all-air duct systems.
- FCU is available in many sizes, including with a self-finish galvanized steel chassis or a painted casing.
- Sound level in fan coil unit low and zones can be individually controlled.
- FCU is very efficient and energy consumption is less.
- Control and maintenance of FCU is also easy.

1.5.2. Disadvantages:

- The FCU requires more maintenance than "all air" systems, and the maintenance work is performed in occupied areas.
- Interior zones may require separate ducts to deliver outside (ventilation) air.

1.6. Air Handling Unit (AHU)

Air handling unit (AHU), is a device used to circulate the air as part of a heating, ventilating, and air-conditioning (HVAC) system. An air handling unit is usually a big metal box having a blower, chambers, heating or cooling elements, dampers and sound attenuators. AHU generally connect to a ductwork ventilation system that allocates the cooled air through the house or rooms and takings it to the AHU.

1.6.1. Air handling components

The main component of AHU is given below:

1.6.1.1. Filters

Air filter is used in the AHU in order to deliver clean dirt-free air to the house occupants. This air filter is generally placed leading in the air handling unit in order to retain all the other apparatuses clean. Depending upon the grade of filtration required, air filters will be organized in two or more successive banks with a coarse-grade section filter provided opposite of a fine-grade bag filter.

1.6.1.2. Heating or cooling elements

Air handlers need to deliver cooling, heating, or both to variation the supply air temperature, and humidity level contingent on the position and requirement. Such conditioning is delivered by a heat exchanger coil. Such coils may be direct or indirect in relation to the medium providing the cooling or heating effect.

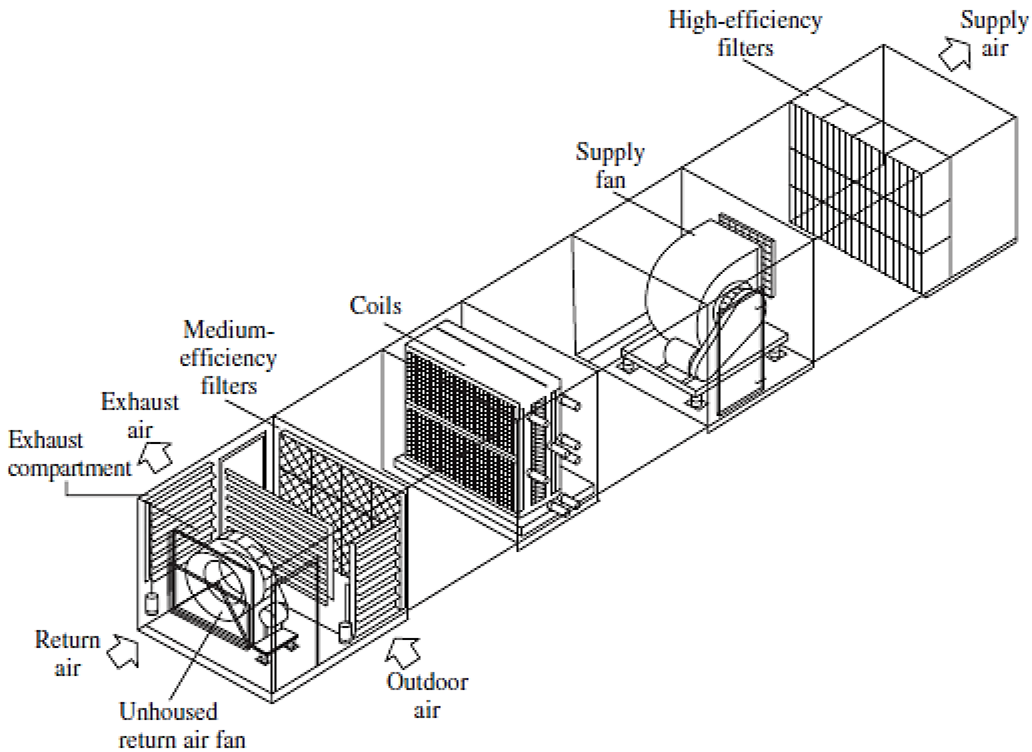

Fig.1.3. Air handling unit

1.6.1.3. Humidifier

Humidification is habitually essential in colder weathers where nonstop warming will make the air drier, resulting in uncomfortable air quality. Generally evaporative type humidifier is used.

1.6.1.4. Blower or fan

Air handlers generally employ a big blower, which is driven by an AC generation electric motor to transfer the air. The blower runs at a constant speed. Flow rate of air can be controlled by inlet blades or outlet dampers on the fan. Numerous blowers are used in big commercial AHU, normally placed at the end of the air handling unit and the opening of the source ductwork.

CHAPTER – 2

LITERATURE REVIEW

G. S. Sharma et al. [1] designed a duct for an air conditioning system in an office building and analyzed the importance of duct design which creates an impact of system performance. Improper duct designs led to problems such as frictional loss, uneven cooling in the building, increased installation cast, increased noise level and power consumption. The above problems highlighted the need for an optimum duct design and effective layout of the duct. The authors used hand calculation and software tool both for designing the duct. They found that the circular duct has a less pressure drop than the rectangular duct. R. Whalley et al. [2] considered HVAC modeling methods for large scale, spatially dispersed systems. In this paper, they discussed existing techniques and proposals for the application of novel analysis. Tengfang T. Xu et al. [3] did field study on the performance of five thermal distributed systems in four large commercial buildings. They studied about the air leakage from duct, and concluded that the air leakage in large commercial systems varied significantly from a system to system. The energy loss due to a leak can be minimized by using duct sealing and duct insulation. Baris Ozerdem et al. [4] studied the energy loss related to the air leakage by using power law model. The measurements were made on different types of duct having different diameter. After measurements, they concluded that the most of the air leakage was from the joint and this air leakage was reduced by about 50% by using sealing gaskets. Michal Krajcík et al. [5] studied experimentally air distribution, ventilation effectiveness and thermal environments, in a simulated room in a low-energy building heated and ventilated by warm air. The measurements were performed at different outdoor conditions, internal heat gain, air change rates. Their study showed that the warm air heating and floor heating system did not affect the significant risk of thermal discomfort. William J. Fisk et al. [6] did field studies in large commercial buildings and they investigated the effective leakage areas ELAs, air-leakage rates, and conduction heat gains of duct systems. Air leakage rates were measured by using different method and their result were compared. They found that the air leakage rate varied from 0% to 30%. Also, heat gains between the cooling coils and the supply registers caused supply air temperatures to increase, on average, by 0.68°C to 28°C. Liping Pang et al. [7] determined the ratio of fresh air to recirculation air. The conditioned temperature of different types of inlets were designed carefully to achieve the high air quality, thermal comfort and energy saving. Furthermore, some experiments were conducted and their performances were compared with the other systems. Their results indicated that, the improved pattern maintain high air quality, because it transported more fresh air directly to the breathing zone and circulated it around the upper body of passengers. K. Srinivasan et al. [8] gained an experience for evaluation of air leakages in components of air conditioning systems by designing and testing of orifice plate-based flow measuring systems. The coefficients of discharge were evaluated and compared with the Stolz equation which value were higher, the deviations being larger in the low Reynolds number. It was observed that a second-degree polynomial was inadequate to relate the pressure drop and flow rate. Huan-Ruei Shiu et al. [9] designed an exhaust duct system using the dynamic programming method in semiconductor factory which considered system pressure equilibrium the least life-cycle cost to originate the duct size and fan capacity. Their results showed that the outcomes value satisfied the requirements on the range of duct diameter. Also, the differences between the design and simulation (actual operation) resulted under DPM were found to be much lower than those of other methods. Wanyu R. Chan et al. [10] analyzed the air leakage measurements of 134,000 single-family detached homes in the US, using normalized leakage. They performed regression analyses to examine the relationship between NL and various house characteristics. Their results indicated that the regression model predicted 90% of US houses had NL between 0.22 and 1.95, with a median of 0.67. Dongliang Zhang et al. [11] studied the energy saving possibility of digital variable multiple air conditioning system and compared to the other the air systems with constant air volume and primary air fan coil system. Their results revealed that the energy saving of DVM air conditioning system was significant under only part load condition and this system was significant when building area was less than 20,000 m^2. A. Gallegos-Muñoz et al. [12] studied the effect in the measurements of flow in air conditioning system caused by fitting. They developed numerical simulation using CFD where the Reynolds Averaged Naviere Stokes equations were solved through an approach of finite volume method using several turbulence models. Their results indicated that the mass flow rate was decreased when no of joints were increased. Also, the work gave information about the behavior of flow measurements made downstream. Isak Kotcioglu et al. [13] found out an optimum value of design parameters in a rectangular duct by using Taguchi method. Their analysis was performed with an optimization process to reach the minimum pressure drop and maximum heat transfer. After some experiments they gave a suitable designed parameter which satisfied the condition i.e. less friction drop, maximum heat transfer. Omer Kaynakli et al. [14] gave a review study to find economic thermal insulation thickness for pipe and ducts with different geometries in various industries. The purpose of their study was to determine the critical thickness insulation for different geometries. The basic result, economic analysis method, heat transfer method, optimization procedure were used for comparison. After that the effective parameters of the optimal thickness were examined. Tabish Alam et al. [15] studied the effect of turbulators for friction characteristic and heat transfer in air ducts. Turbulators were used to improve the performance of air heater and heat exchanger. The relationship was presented in terms of non-dimensional parameter for friction factor and heat transfer in air duct. Also they examined heat transfer increase and flow structure in air ducts. Kwang Ho Lee et al. [16] presented a paper for thermal decay in under floor air distribution (UFAD) systems. They used an energy simulation program, Energy plus to explain the fundamental of thermal decay, energy consumption and parameter that affected the thermal decay. Using the UFAD, they found that the system has several advantages like improved thermal comfort and indoor air quality, reduced life cycle, and improved energy efficiency. Also the temperature rise was not significantly affected in thermal decay. Megan A. Bos et al. [17] did a field study of thermal comfort with under floor air distribution. The field study was conducted between summer and winter climate. They selected 100 male and 100 female participants for the survey. According to their study, 50 % told that slightly cooler than neutral, 21 % told that warmer than neutral, 20 % indicated that they would prefer more air movement, remaining 9 % responded that they experienced some discomfort. Miroslaw Zukowski [18] gave an experimental data for forced convection heat transfer in a heat exchanger. The experiments were conducted to study the pressure drop characteristics with different geometries of air flow. The material used for heat exchanger was polyvinyl chloride (PVC). The experimental value given in that paper provided useful information for design of an UFAD in residential building. Andrew Kusiak et al. [19] presented a model for minimizing the cooling output of an air handling unit by data driven optimizing approach. For analyzing, the dynamic model was studied which built by four different data mining algorithms. The evolutionary strategy algorithm was used to solve the optimization problem. A bi-objective optimization model was proposed to minimize the cooling output for maintaining the thermal properties of the supply air within a range. The result showed that the cooling output was reduced, when the supply air temperature and humidity in the AHU is in an adequate range. P. Jaboyedoff et al. [20] studied the energy use in AHU's and suggested several energy efficient methods to deliver good indoor air and also effective indoor environment conditioning. They studied to identify the pollution sources in AHU's and characterized these sources using measurement protocols and to propose measures to avoid this pollution. The results showed that, in most temperate environments, naturally ventilated buildings could be kept colder during the cooling season than the AC buildings. Gang Wang et al. [21] presented an AHU system energy modeling, supply air temperature optimization, control sequence development and simulated energy savings. Economizer cycle was used in the AHU system for a free cooling under certain outside air condition. When outside air was dry, the temperature of outside air is high because of that space might have less cooling effect. So the higher air temperature was required to produce cooling effect and reduce terminal box reheat. Increase air temperature led to increase air flow as well as fan power. To minimize this energy consumption optimization question is formed in which optimal supply air temperature was identified. Andrew Kusiak et al. [22] were used data mining approach to reduce the energy consumption of an air handling units. They developed a non-linear model to minimize energy consumption, maintain supply air temperature and static pressure in a predetermined range. A dynamic, penalty-based algorithm was designed to solve the proposed model. They used 200 test data point to validate the proposed model and their result showed that the energy consumption by the AHUs was reduced by about 23%. Jose Fernandez-Seara et al. [23] presented the experimental analysis on pressure drop and heat transfer of a fan-coil unit with ice slurry as a coolant. The ice slurry was produced from an ethylene glycol and 10% (by wt) of aqueous solution. The fan-coil capacity was experimentally determined for chilled water and melting ice slurry with different inlet ice fractions considering in each case three different fan rotation velocities. The heat transfer experimental results showed that the air side thermal resistance controlled the heat transfer process ranging from 80% to 88% of the overall thermal resistance. M.T. Ke [24] designed test method for variable air volume (VAV) fan coil units to establish complete testing and rating procedures. Using the ASHRE standard 79-1984, for a constant air volume (CAV) they proposed a testing method by considering VAV features and recognized a testing platform for VAV fan coil units. The difference between the variable air volume and constant air volume fan coil unit result were differentiated. After differentiation they concluded that the VAV fan coil units had the energy efficient method.

Francois Remi Carrie et al. [25] performed a field study on 42 duct systems in Belgium and France to investigate the implication of duct leakage. Their study reported that the leakage rate appeared to be three times greater than the maximum allowed leakage adopted in EUROVENT. Because of this leakage the overall effectiveness of the system reduces. Also, the ductwork air tightness affected the duct leakage. The saving energy in duct was calculated at the European level based on estimates of the number of buildings equipped with the mechanical ventilation systems. K.W. Cheong [26] proposed a new technique to measure the air flow rate, velocity in air distribution system. In present day, vane anemometer and pitot-tube are used to measure the air velocity at supply diffuser and ducts. This process is slow and also some error can occur during measurement. By using the tracer-gas technique, pitot static transverse method air flow has measured for 300*300 mm^2 duct. After comparison, their result shows that tracer gas technique has better accuracy as compared to the pitot-static transverse method. Also, it is simple and accurate method. Sergio Marinetti et al. [27] analyzed the air speed distribution inside the duct and its effect on heat exchanger performance by using stereoscopic particle image velocimetry. Two configuration of duct, in which the air was forced by one and two fans, were tested. Experimental flow field, upstream and downstream of the evaporator were calculated at three different heights. Their results showed that, in two fan configuration, about 7% reductions in the thermal energy losses was noticed as compared with the single fan configuration. Maria Justo Alonso et al. [28] performed the analysis of the tunnel design by CFD simulations with the ANSYS Airpak software tool. The analysis was done in an already existing freezing plant where different ceiling geometries were tested in order to improve the design, identify problem area and obtain to better air distribution. The simulation result showed that the fan power and air distribution could be strongly influenced by ceiling design. The improvement in design gave a more homogeneous flow which led to the improvement in the air quality. Because of this reduction in energy use and freezing time which might be economically beneficial. Cuimin Li et al. [29] developed a new technology to analyze the indoor air temperature for gravity air conditioning. By experimentation, they investigated vertical and horizontal temperature distribution. Their results indicated that the temperature was in the range of 23°C -28°C in the working area, which could meet the cooling requirement. Their results also indicated that for the higher temperature of upper zone, gravity air conditioning was more energy efficient as compared to traditional technique.

CHAPTER – 3

METHODOLOGY

This project gives the fundamental principles of duct or air distribution system design for a multi-story building. There are mainly three types of duct sizing method namely (i) equal friction method, (ii) modified friction method (static regain method) and (iii) velocity reduction method. Now a days, the use of manual duct calculator is normal and computer aided duct design is becoming more popular. Also understanding the friction chart is very important to use this manual duct calculator, because these are the foundations of the other methods. This will provide the necessary knowledge to the duct design error and overcome to the errors.

For designing a proper duct system, it is necessary to estimate cooling load which is used to select the zone and air flow rate that the duct system distributes. Once the air flow rate is determined, the duct system component can be placed. This includes the supply and returns diffusers and decides to air handling unit (AHU) or fan coil unit (FCU) is good for that space.

3.1. General rules for duct design

- Air should be conveyed as directly as possible to economize on power, material and shape.
- Sudden change in direction should be avoided.
- Air velocities in ducts should be within the permissible limits to minimize losses.
- Rectangular ducts should be made as nearly square as possible. This will ensure minimum ducts surface. An aspect ratio of less than 4:1 should be maintained.
- Damper should be provided in each branch outlet for balancing the system.

3.2. Duct Design Criteria

Many factors are considered when designing a duct system. They are as follows

1. Space availability
2. Installation cost
3. Air friction loss

4. Noise level
5. Duct heat transfer and airflow leakage

3.2.1. Space Availability

The sizing of a duct depends on the space available in the building. Ceiling plenums, duct chases, obstruction like walls and beam dictate the size of duct to be used, irrespective of the size at a least cost. At the time of design, the duct coordination is required to avoid sprinkler piping, power and light fixtures. For this, header duct and runouts are easier to locate. Larger the trunk and branch ducts greater the coordination required with equally large piping.

3.2.2. Installation Cost

While designing, the duct installation cost is very important. This includes size of ducts, type of material used, number and complexity of the duct fitting and height of the site conditions impacting duct installation labor. Use least no of fitting as possible to lower the installation cost.

3.2.3. Air Friction Loss

Air friction loss is affected mainly by the duct size and shape, the material used, fittings used. According to "Carrier Handbook" round galvanized sheet metal has the lowest friction loss per meter, while the flexible ductwork has the highest friction loss per meter. The quality of fitting has a direct effect on the overall pressure drop of a duct system, smooth and efficient fitting with a low turbulence reduce the duct system air pressure drop. A direct route using round duct with less fitting and size changes can have a less friction loss in comparison with the similar size rectangular system with a longer route and size changes at each branch duct.

3.2.4. Noise Level

The modern AC systems require control of noise level below a particular level in addition to the control of humidity, temperature and air velocity of excessive noise which causes uncomfortable feeling. The equipment as blowers, humidifiers, motors and many others contribute noise to the air conditioned space. The air passing through the ducts and grills also create noise.

3.2.5. Heat Transfer and Leakage

Ductwork that runs through very warm or cold areas can suffer heat gain or loss that effectively reduce the capacity of the cooling and heating equipment, result in occupant discomfort and higher operating cost. Leakage in duct also affects the capacity of cooling equipment and may create odors.

3.3. Pressure in Duct

The flow of air within a duct system is produced by the pressure difference existing between the different locations. The greater the pressure difference, the faster the air will flow. The following are the three types of pressures involved in a duct system.

3.3.1. Static Pressure (P_s)

The static pressure always exists in a duct system. The pressure which is independent upon the air movement called static pressure. This type of pressure pushes against the wall of the duct. It tends to rush a duct when its force is greater than that of atmospheric pressure and tends to collapse when its force is less than that of the atmosphere. These pressures overcome the friction and shock losses as the air is flow.

3.3.2. Velocity Pressure (P_v)

The velocity or dynamic pressure is equal to the drop in static pressure necessary to produce a given velocity of flow. In other words, it is equal to the increase of static pressure possible when velocity is reduced to zero.

3.3.3. Total Pressure (P_t)

It is the algebraic sum of the static pressure and dynamic pressure.

$$P_{t=} = P_s + P_v \qquad \text{………………………} \; (3.1)$$

P_t = total pressure, Pa

P_s = static pressure, Pa (measured by any pressure measuring instrument)

P_v = velocity pressure

$= \frac{\rho V^2}{2}$, (for air ρ = 1.024 kg/m3)

$$=0.602V^2 \qquad \text{……………………………} \; (3.2)$$

V = fluid mean velocity, m/s

$$= \frac{Q}{A} \qquad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots \text{(3.3)}$$

Where, Q = air flow rate, m^3/sec

A = cross sectional area, m^2

3.4. Pressure Losses in Ducts

Pressure is lost due to friction between the moving particle of the fluid and the interior surfaces of a duct. When the pressure loss occurs in a straight duct, then this loss is known as friction loss. The pressure loss is due to the changes of direction of air flow such as bends, elbows etc. and at the change of cross section of the duct, this loss is known as dynamic losses.

3.4.1. Pressure Loss due to Friction in Ducts

The pressure loss due to friction in ducts may be obtained by using the Darcy's formula, i.e.

$$p_f = \frac{fL\rho_a V^2}{2D_h} \qquad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots \text{(3.4)}$$

Where

P_f = pressure loss due to friction in N/m^2

L = length of the duct in meters

f = friction factor depending upon the surface of the duct

ρ_a = density of air in kg/m^3

V = mean velocity of the air flowing through the duct in m/s

D_h= hydraulic diameter in m

$$= \frac{\text{cross sectional area of the duct (A)}}{\text{perimeter of the duct (P)}}$$

$= \frac{D}{4}$ for circulation cross section, where D is a diameter of duct

$= \frac{ab}{2(a+b)}$, where a and b is a side of rectangle

The value of friction factor (*f*) for different Reynolds numbers and different roughness factor find directly from the Moody chart as shown in Fig. 3.1.

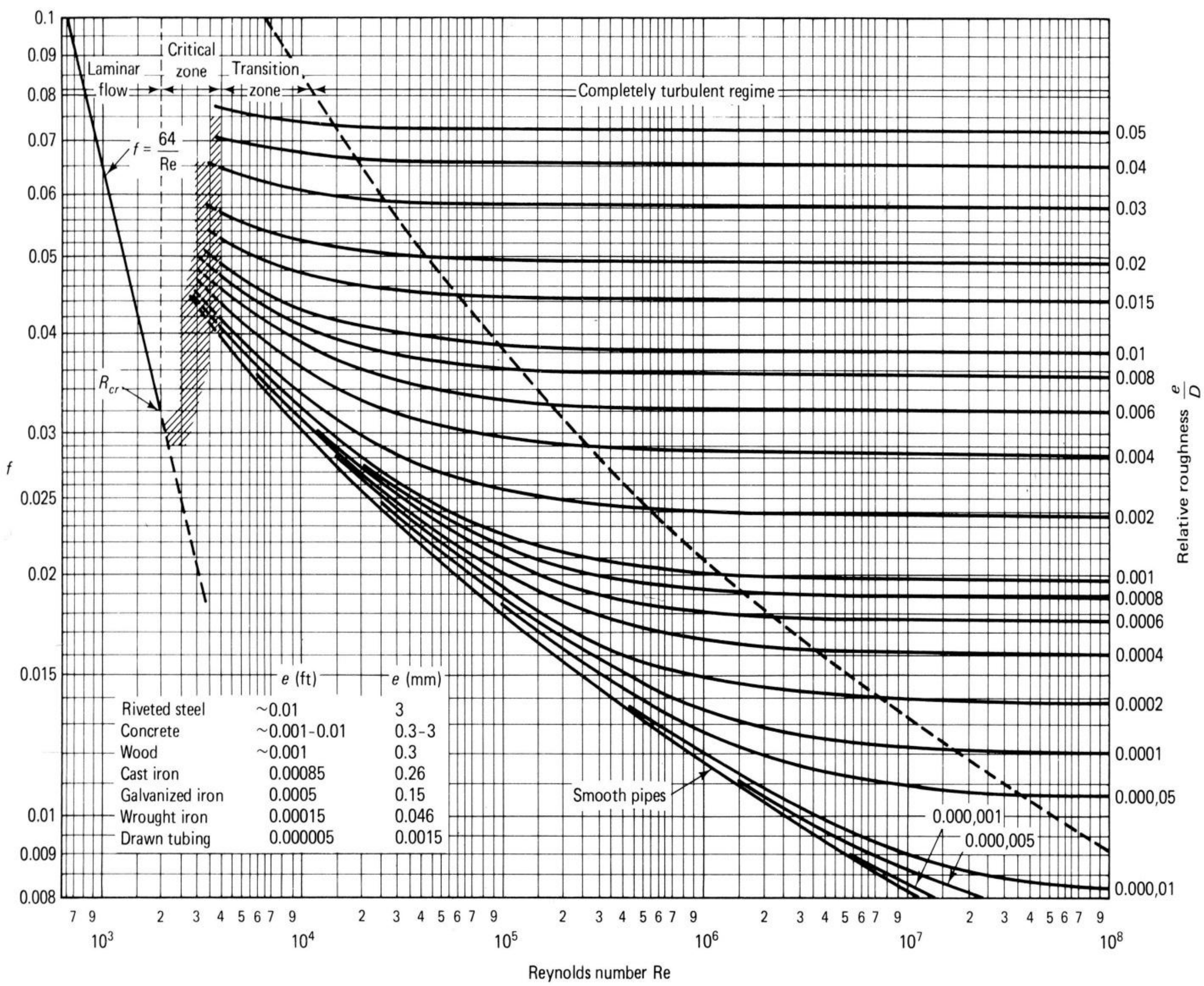

Figure 3.1. Moody Chart

3.4.2. Dynamic Losses in Ducts

The dynamic losses are caused due to the change in direction or magnitude of velocity of the fluid in the duct. The change in the direction of the velocity occurs at bends and elbow. The change in the magnitude of velocity occurs when the area of duct changes i.e. enlargement, contraction, suction etc.

The dynamic pressure loss Δp_d is proportional to the velocity pressure and it is expressed as a product of the downstream velocity pressure p_v and a dynamic loss coefficient (K).

$$\Delta p_d = Kp_v = K\left(\frac{\rho C^2}{2}\right)$$

Where, V = downstream velocity.

The losses in elbows, fittings etc. are also expressed in terms of an equivalent length L_e of the duct, so that

$$\Delta p_d = Kp_v = \left(\frac{4\,fL_e\,p_v}{D}\right) \qquad \text{.........................} \ (3.5)$$

3.5. Friction Chart

The frictional pressure loss for circular ducts (in mm of water) for various velocities (in m/s) and duct diameters (in mm) obtained directly from the friction chart as shown in Fig. 3.2. In this chart, the vertical ordinate represent volume flow rate of air in m^3/s and the horizontal ordinate represents frictional pressure loss in mm of water per unit length of the circular duct. These charts are valid for 20°C and 1.013 bar and clean galvanized iron ducts with joints and seams [27].

3.6. Duct Velocity Ranges

The velocities in the ducts must be high enough to reduce the size of the ducts but it should be low enough to reduce the noise and pressure losses to economize power requirement. The velocities recommended for various applications are given in Table 3.1 [27]:

Table 3.1. Recommended Velocities in (m/min)

Designation	Residences	School, theatres and public building	Industrial building
Outdoor air intake	150	150	150
Filters	75	90	105
Cooling coil	135	150	180
Air washer	150	150	150
Fan outlet	300 – 480	400 – 600	480 – 725
Grills	40 – 60	60 – 80	80 – 100

Main duct	210 - 300	300 – 400	360 – 540
Branch duct	180	180 – 270	240 – 300
Branch riser	150	180 – 210	240

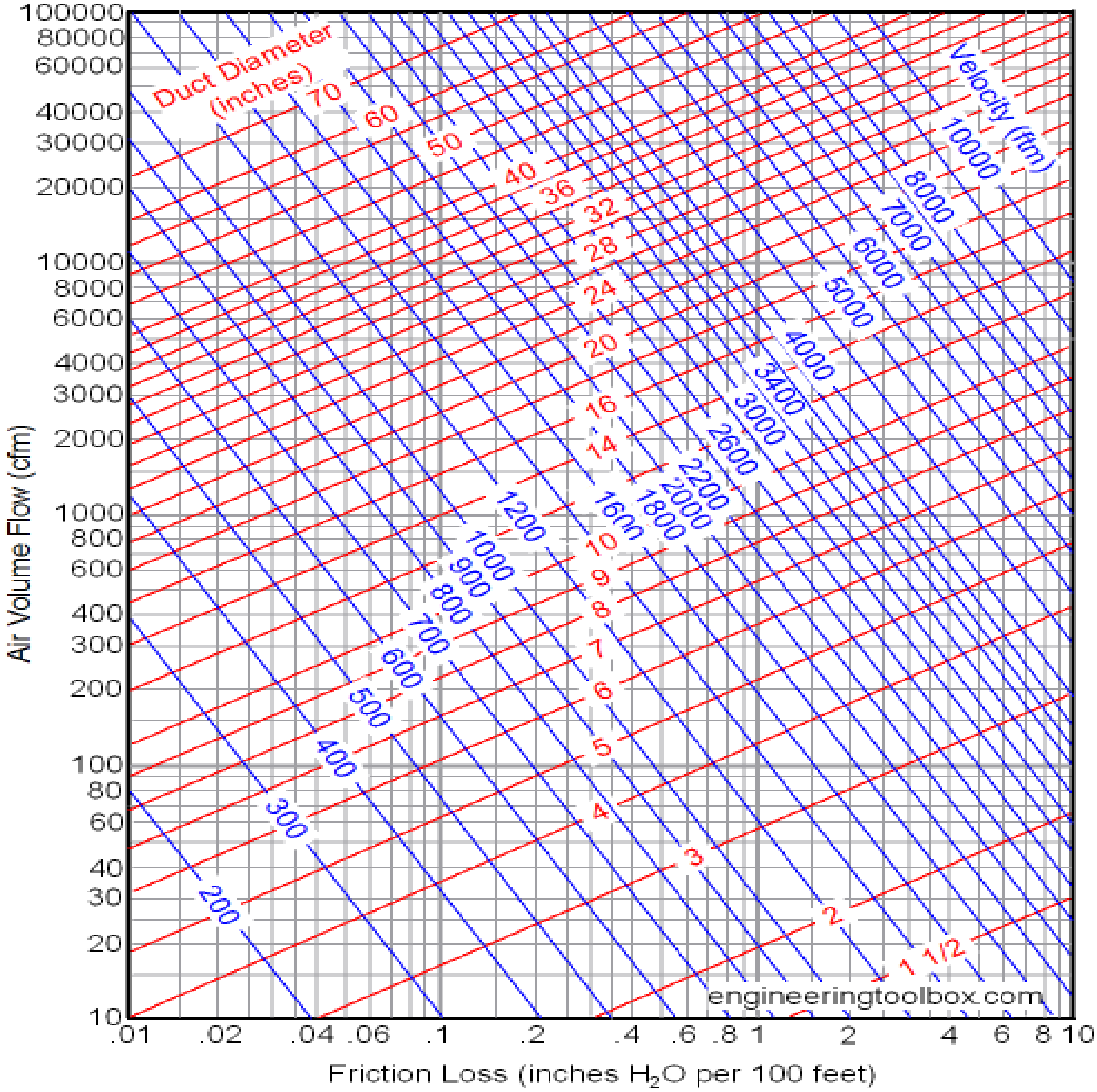

Figure 3.2.- Duct Friction Chart

3.7. Duct Material Roughness

Duct material roughness refers to the inside surface of the duct material the rougher the surface, higher the friction loss. The recommended roughness for different material pipes or ducts are given in Table 3.2 [27].

Table 3.2. The recommended roughness for different material pipes or ducts material

Types of duct or pipes material	Roughness category	Absolute roughness
PVC plastic pipe	Smooth	0.01 – 0.05
Galvanized steel longitudinal seam	Medium smooth	0.05 - 0.1
Galvanized steel continuous roll	Medium smooth	0.06 – 0.12
Fibrous glass duct, rigid	Medium rough	0.9
Flexible duct metallic	Rough	1.2 – 2.1
Aluminum	Smooth	0.04 – 0.06
Concrete	Medium	1.2
Concrete	Smooth	0.3
Commercial steel pipe	Smooth	0.045

3.8. Equivalent Duct Diameter

In order to find the equivalent diameter of a circular duct for a rectangular duct for the same pressure loss per unit length, Huebscher developed a relationship between rectangular and round duct. According to this,

$$D_e = \frac{1.30(ab)^{0.625}}{(a+b)^{0.250}} \qquad \text{.............................. (3.7)}$$

Where,

D_e = equivalent circular diameter of rectangular duct for equal length, mm

a = length one side of duct, mm

b = length adjacent side of duct, mm

Equivalent round duct diameter can also be determined by using Fig. 3.3 which is based on the above equation [26].

Lgth Adj.[b]	Length of One Side of Rectangular Duct (a), mm																			
	100	125	150	175	200	225	250	275	300	350	400	450	500	550	600	650	700	750	800	900
	Circular Duct Diameter, mm																			
100	109																			
125	122	137																		
150	133	150	164																	
175	143	161	177	191																
200	152	172	189	204	219															
225	161	181	200	216	232	246														
250	169	190	210	228	244	259	273													
275	176	199	220	238	256	272	287	301												
300	183	207	229	248	266	283	299	314	328											
350	195	222	245	267	286	305	322	339	354	383										
400	207	235	260	283	305	325	343	361	378	409	437									
450	217	247	274	299	321	343	363	382	400	433	464	492								
500	227	258	287	313	337	360	381	401	420	455	488	518	547							
550	236	269	299	326	352	375	398	419	439	477	511	543	573	601						
600	245	279	310	339	365	390	414	436	457	496	533	567	598	628	656					
650	253	289	321	351	378	404	429	452	474	515	553	589	622	653	683	711				
700	261	298	331	362	391	418	443	467	490	533	573	610	644	677	708	737	765			
750	268	306	341	373	402	430	457	482	506	550	592	630	666	700	732	763	792	820		
800	275	314	350	383	414	442	470	496	520	567	609	649	687	722	755	787	818	847	875	
900	289	330	367	402	435	465	494	522	548	597	643	686	726	763	799	833	866	897	927	984
1000	301	344	384	420	454	486	517	546	574	626	674	719	762	802	840	876	911	944	976	1037
1100	313	358	399	437	473	506	538	569	598	652	703	751	795	838	878	916	953	988	1022	1086
1200	324	370	413	453	490	525	558	590	620	677	731	780	827	872	914	954	993	1030	1066	1133
1300	334	382	426	468	506	543	577	610	642	701	757	808	857	904	948	990	1031	1069	1107	1177
1400	344	394	439	482	522	559	595	629	662	724	781	835	886	934	980	1024	1066	1107	1146	1220
1500	353	404	452	495	536	575	612	648	681	745	805	860	913	963	1011	1057	1100	1143	1183	1260
1600	362	415	463	508	551	591	629	665	700	766	827	885	939	991	1041	1088	1133	1177	1219	1298
1700	371	425	475	521	564	605	644	682	718	785	849	908	964	1018	1069	1118	1164	1209	1253	1335
1800	379	434	485	533	577	619	660	698	735	804	869	930	988	1043	1096	1146	1195	1241	1286	1371
1900	387	444	496	544	590	663	674	713	751	823	889	952	1012	1068	1122	1174	1224	1271	1318	1405
2000	395	453	506	555	602	646	688	728	767	840	908	973	1034	1092	1147	1200	1252	1301	1348	1438
2100	402	461	516	566	614	659	702	743	782	857	927	993	1055	1115	1172	1226	1279	1329	1378	1470
2200	410	470	525	577	625	671	715	757	797	874	945	1013	1076	1137	1195	1251	1305	1356	1406	1501
2300	417	478	534	587	636	683	728	771	812	890	963	1031	1097	1159	1218	1275	1330	1383	1434	1532
2400	424	486	543	597	647	695	740	784	826	905	980	1050	1116	1180	1241	1299	1355	1409	1461	1561
2500	430	494	552	606	658	706	753	797	840	920	996	1068	1136	1200	1262	1322	1379	1434	1488	1589
2600	437	501	560	616	668	717	764	810	853	935	1012	1085	1154	1220	1283	1344	1402	1459	1513	1617
2700	443	509	569	625	678	728	776	822	866	950	1028	1102	1173	1240	1304	1366	1425	1483	1538	1644
2800	450	516	577	634	688	738	787	834	879	964	1043	1119	1190	1259	1324	1387	1447	1506	1562	1670
2900	456	523	585	643	697	749	798	845	891	977	1058	1135	1208	1277	1344	1408	1469	1529	1586	1696

Figure 3.3. Equivalent round duct diameter

3.9. Duct Design Method

There are mainly three methods which are commonly used for duct design. These are:

3.9.1. Velocity Reduction Method

The duct are designed in such a way that the velocity decreases as flow proceeds. The pressure drops are calculated for these velocities for respective branches and main duct. The pressure at the outlet is adjusted by damper in the respective ducts.

The advantages of this system are given below:

- This method is the easiest among all methods in sizing the duct diameters.
- The velocities can be adjusted to avoid noise.
- This is adopted only for simple system.

The major disadvantage of this system is that, considerable judgment is required in selecting velocities to make the system optimum in economy and power.

3.9.2. Equal Friction Drop (friction loss) Method

In this method, the size of the duct is decided to give equal pressure drop per meter length an all ducts. The velocities are automatically reduced in the branch duct as the flow is decreased.

The main advantage of this method is that, if the duct layout is symmetrical giving the same length in each run, then no dampers are required to balance the system as this method gives equal pressure loss in various branches.

Disadvantages of this method is that, if the runs are of different lengths, then the shortest run will have a minimum drop and air will come out with higher pressure compared with long run ducts. It is necessary to reduce this high pressure of coming out air with the help of damper or high velocity can be reduced in a shorter run, but high velocity may create an objectionable noise. Therefore noise absorbing outlets must be provided.

3.9.3 The Static Regain Method

For the perfect balancing of the air duct layout system, the pressure at all outlets must be made same. This can be done by equalizing the pressure losses in the various branches. This is possible if the friction loss in each run is made equal to pressure gain due to reduction in velocity. The gain in pressure due to change in velocity is given by

$$SPR = R\left(\frac{v_1^2 - v_2^2}{2g}\right) \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (3.8)$$

Where,

SPR = static pressure regain

R = static regain factor

The advantages of this system are

- It is possible to design long runs as well as short runs for complete regain.
- It is sufficient to design the main duct for complete regain and use the same pressure at outlets of the branches.

The disadvantages of this system are

- This method allows for balancing but reducing velocity increases the duct size and it should not exceed the economic limit.

CHAPTER – 4

DESIGN CALCULATION

In the TIIR building there are total 18 rums, where cooling is required. The list of rooms in floor wise in the TIIR Building and where cooling load is required is given below:

TIIR BUILDING						
GROUND FLOOR						
S.N.	**Room/ Hall**	**Width (m)**	**Length (m)**	**Area (m^2)**	**Celling Ht (m)**	**AC Requirement**
1	120 seat Lecture room	14.17	8.67	122.85	3.40	122.85
2	Direct TIIR	3.57	6.87	24.53	3.40	24.53
3	Admin Office	5.27	6.87	36.20	3.35	36.20
4	Placement office	2.97	6.87	20.40	3.40	20.40
5	IPR Office	5.27	6.87	36.20	3.35	36.20
6	Professor	3.57	6.87	24.53	3.40	24.53
7	120 seat Lecture room	14.17	8.67	122.85	3.40	122.85
8	Office room	12.27	6.97	85.52	3.40	85.52
9	Meeting room	12.27	6.97	85.52	3.40	85.52
10	Central Workshop	14.17	19.97	282.97		
11	Library	6.97	6.47	45.10	3.32	45.10
12	Dining	9.67	6.97	67.40	3.32	67.40
13	Alumini Relation	6.97	6.97	48.58	3.32	48.58
14	Alumini Visitors	6.97	12.74	88.80	3.32	88.80
15	Placement Cell	14.17	19.97	282.97		

FIRST FLOOR						
1	Common Facilities	14.04	20.09	282.06		
2	Interview Room	5.23	6.97	36.45	3.35	36.45
3	Interview Room	9.07	6.97	63.22	3.35	63.22
4	Working Modules	14.04	8.77	123.13		
5	Working Modules	12.53	6.97	87.33		
6	Seminar Room	12.53	6.97	87.33	3.35	87.33
7	Placement Cell	14.04	20.08	281.92		
8	Working Modules	6.97	6.66	46.42		

9	Working Modules	6.97	6.00	41.82		
10	Working Modules	6.97	6.97	48.58		
11	Working Modules	6.97	6.97	48.58		
12	Working Modules	6.97	6.97	48.58		
13	Working Modules	6.97	6.00	41.82		
14	Working Modules	6.97	6.66	46.42		
15	Central Design Office	14.17	20.09	284.68	3.32	284.68
16	Auditorium	20.00	25.00	500.00	7.55	500.00

SECOND FLOOR						
1	Working Modules	14.17	8.67	122.85		
2	Library Facilities	9.07	6.87	62.31	3.35	62.31
3	Working Modules	9.07	6.87	62.31		
4	Working Modules	14.17	8.67	122.85		
5	Working Modules	12.27	6.97	85.52		
6	Working Modules	12.27	6.97	85.52		
7	Working Modules	14.17	19.97	282.97		
8	Working Modules	6.97	6.43	44.82		
9	Working Modules	6.97	6.00	41.82		
10	Working Modules	6.97	6.97	48.58		
11	Working Modules	7.20	6.97	50.18		
12	Working Modules	9.44	6.97	65.80		
13	Working Modules	6.97	6.00	41.82		
14	Working Modules	6.97	6.66	46.42		
15	Working Modules	14.17	19.97	282.97		

On the basis of cooling load required in a room, fan coil unit (FCU) or air handling unit (AHU) is used. In this work it is decided to use FCU where cooling load required up to 5 tons and to use AHU above 5 tons. As we know that, for FCU there is no duct is required. So, we calculate the duct size only for those rooms where cooling load is required more than 5 tons or where AHU is used. For that purpose firstly calculate the air flow rate/dehumidified air. After that the calculation for duct dimension has to be done.

Calculation for dehumidified air quantity

Room Rise = (1 – by pass factor)*(Room temp - ADP)

Dehumidified Air = RSH / (20.44 * dehumidified rise)

Where,

ADP = apparatus due point

RSH = room sensible heat

Calculation for duct size/dimension:

1. First find out the air flow rate i.e. dehumidified air and cooling load.
2. Based on cooling load select AHU or FCU which is to be installed. For FCU there is no need to duct system. If AHU then calculate the duct dimension.
3. Select initial velocity (from table 3.1)
4. Duct area = $\frac{\text{air flow rate}}{\text{velocity}}$
5. Select duct size/dimension (From figure 3.3), also Equivalent duct diameter.
6. Then initial friction rate is determined by using friction chart, on the basis of air quantity and equivalent duct diameter or velocity of air (figure 3.2).

The ducts size room by room are calculated and given below:

1. 120 seat lecture room 1

By pass factor = 0.12

Room temperature = 23°C

ADP = 9°C

RSH = 27078.65 W

Room Rise = (1 – 0.12)*(296 - 282)

= 12.32

Dehumidified Air = 27078.65 / (20.44 * 12.34)

= 107.53 m^3/min

Safety factor (5%) = 5.37 m^3/min

Total dehumidified air = 112.90 m^3/min ≈ 113 m^3/min

Cooling load = 12.39 tons

Initial velocity = 300 m/min

Duct area = $\frac{\text{air flow rate}}{\text{velocity}}$

$= \frac{113}{300} = 0.38\ m^2 = 4.09\ ft^2$

Duct size = 26 * 24 inch = 650 * 600 mm

Equivalent duct diameter = 27.2 inch = 680 mm

Friction rate = 0.0514

2. Direct TIIR

By pass factor = 0.12

Room temperature = 23°C

ADP = 9°C

RSH = 5087.24 W

Room Rise = (1 – 0.12)*(296 - 282)

= 12.32

Dehumidified Air = 5087.24 / (20.44 * 12.32)

= 20.20 m^3/min

Safety factor (5%) = 1.01 m^3/min

Total dehumidified air = 21.21 m^3/min ≈ 21 m^3/min

Cooling load = 2.11 tons

In direct TIIR cooling load is less than 5 tons. So, there is no need to design the duct. Here FCU is installed.

3. Admin Office

By pass factor = 0.12

Room temperature = 23°C

ADP = 10°C

RSH = 7133.79 W

Room Rise = (1 – 0.12)*(296 - 283)

= 11.44

Dehumidified Air = 7133.79 / (20.44 * 11.44)

= 30.50 m^3/min

Safety factor (5%) = 1.52 m^3/min

Total dehumidified air = 32.02 m^3/min ≈ 32 m^3/min

Cooling load = 3.03 tons

In admin office cooling load is less than 5 tons. So, there is no need to design the duct. Here FCU is installed.

4. Placement office

By pass factor = 0.12

Room temperature = 23°C

ADP = 9°C

RSH = 5901.09 W

Room Rise = (1 – 0.12)*(296 - 282)

= 12.34

Dehumidified Air = 5901.09 / (20.44 * 12.34)

= 23.43 m^3/min

Safety factor (5%) = 1.17 m^3/min

Total dehumidified air = 24.6 m^3/min ≈ 25 m^3/min

Cooling load = 2.27 tons

In placement office cooling load is less than 5 tons. So there is no need to design the duct. Here FCU is installed.

5. IPR Office

By pass factor = 0.12

Room temperature = 23°C

ADP = 10°C

RSH = 7059.59 W

Room Rise = (1 – 0.12)*(296 - 283)

= 11.44

Dehumidified Air = 7059.59 / (20.44 * 11.44)

= 30.12 m^3/min

Safety factor (5%) = 1.5 m^3/min

Total dehumidified air = 31.62 m^3/min ≈ 32 m^3/min

Cooling load = 2.93 tons

In IPR office cooling load is less than 5 tons. So there is no need to design the duct. Here FCU is installed.

6. 120 seat Lecture room 2

By pass factor = 0.12

Room temperature = 23°C

ADP = 9°C

RSH = 27078.65 W

Room Rise = (1 – 0.12)*(296 - 282)

= 12.32

Dehumidified Air = 27078.65 / (20.44 * 12.32)

= 107.53 m^3/min

Safety factor (5%) = 5.38 m^3/min

Total dehumidified air = 112.91 m^3/min ≈ 113 m^3/min

Cooling load = 12.39 tons

Initial velocity = 300 m/min

Duct area = $\frac{\text{air flow rate}}{\text{velocity}}$

$= \frac{113}{300} = 0.38\ m^2 = 4.09\ ft^2$

Duct size = 26 * 24 inch = 650 * 600 mm

Equivalent duct diameter = 27.2 inch = 680 mm

Friction rate = 0.0514

7. Office room

By pass factor = 0.12

Room temperature = 23°C

ADP = 9°C

RSH = 16561.13 W

Room Rise = (1 – 0.12)*(296 - 282)

= 12.32

Dehumidified Air = 16561.13 / (20.44 * 12.32)

= 65.76 m^3/min

Safety factor (5%) = 3.28 m^3/min

Total dehumidified air = 69.04 m^3/min ≈ 69 m^3/min

Cooling load = 6.96 tons.

Initial velocity = 300 m/min

Duct area = $\frac{69}{300} = 0.23\ m^2 = 2.48\ ft^2$

Duct size = 24 * 16 inch = 600 * 400 mm

Equivalent duct diameter = 21.3 inch = 530 mm

Friction drop = 0.0466

8. Meeting room

By pass factor = 0.12

Room temperature = 23°C

ADP = 9°C

RSH = 15923.21 W

Room Rise = (1 – 0.12)*(296 - 282)

= 12.32

Dehumidified Air = 15923.21 / (20.44 * 12.32)

= 63.23 m^3/min

Safety factor (5%) = 3.16 m^3/min

Total dehumidified air = 66.39 m^3/min ≈ 66 m^3/min

Cooling load = 6.77 tons

Initial velocity = 300 m/min

Duct area = $\frac{66}{300}$ = 0.22 m^2 = 2.37 ft^2

Duct size = 20 * 18 inch = 500 * 450 mm

Equivalent duct diameter = 20.7 inch = 517.5 mm ≈ 520 mm

Friction drop = 0.0744

9. Library

By pass factor = 0.12

Room temperature = 23°C

ADP = 9°C

RSH = 10343.02 W

Room Rise = (1 – 0.12)*(296 - 282)

= 12.32

Dehumidified Air = 10343.02 / (20.44 * 12.32)

= 41.07 m^3/min

Safety factor (5%) = 2.05 m^3/min

Total dehumidified air = 43.12 m^3/min ≈ 43 m^3/min

Cooling load = 4.53 tons

In library cooling load is less than 5 tons. So there is no need to design the duct. Here FCU is installed.

10. Alumini Relation

By pass factor = 0.12

Room temperature = 23°C

ADP = 9°C

RSH = 11830.62 W

Room Rise = (1 – 0.12)*(296 - 282)

= 12.32

Dehumidified Air = 11830.62 / (20.44 * 12.32)

= 46.98 m^3/min

Safety factor (5%) = 2.34 m^3/min

Total dehumidified air = 49.32 m^3/min ≈ 49 m^3/min

Cooling load = 4.68 tons

In Alumini relation cooling load is less than 5 tons. So there is no need to design the duct. Here FCU is installed.

11. Dining Room

By pass factor = 0.12

Room temperature = 23°C

ADP = 10°C

RSH = 13965.92 W

Room Rise = (1 – 0.12)*(296 - 283)

= 11.44

Dehumidified Air = 13965.92 / (20.44 * 11.44)

= 59.72 m^3/min

Safety factor (5%) = 2.98 m^3/min

Total dehumidified air = 62.7 m^3/min ≈ 63 m^3/min

Cooling load = 5.85 tons

Initial velocity = 300 m/min

Duct area = $\frac{63}{300}$ = 0.21 m^2 = 2.26 ft^2

Duct size = 22 * 16 inch = 550 * 400 mm

Equivalent duct diameter = 20.4 inch = 510 mm

Friction drop = 0.058

12. Alumini Visitors

By pass factor = 0.12

Room temperature = 23°C

ADP = 10°C

RSH = 16423.45 W

Room Rise = (1 – 0.12)*(296 - 283)

= 11.44

Dehumidified Air = 16423.45 / (20.44 * 11.44)

= 70.23 m^3/min

Safety factor (5%) = 3.51 m^3/min

Total dehumidified air = 73.74 m^3/min ≈ 74 m^3/min

Cooling load = 7.55 tons

Initial velocity = 300 m/min

Duct area = $\frac{74}{300}$ = 0.25 m^2 = 2.69 ft^2

Duct size = 22 * 20 inch = 550 * 500 mm

Equivalent duct diameter = 22.9 inch = 572 mm ≈ 570 mm

Friction drop = 0.058

13. Interview Room 1

By pass factor = 0.12

Room temperature = 23°C

ADP = 10°C

RSH = 7003.83 W

Room Rise = (1 – 0.12)*(296 - 283)

= 11.44

Dehumidified Air = 7003.83 / (20.44 * 11.44)

= 29.95 m^3/min

Safety factor (5%) = 1.49 m^3/min

Total dehumidified air = 31.44 m^3/min ≈ 31 m^3/min

Cooling load = 2.95 tons

In interview room 1 cooling load is less than 5 tons. So there is no need to design the duct. Here FCU is installed.

14. Interview Room 2

By pass factor = 0.12

Room temperature = 23°C

ADP = 9°C

RSH = 10607.05 W

Room Rise = (1 – 0.12)*(296 - 282)

= 12.32

Dehumidified Air = 10607.05 / (20.44 * 12.32)

= 42.12 m^3/min

Safety factor (5%) = 2.10 m^3/min

Total dehumidified air = 44.22 m^3/min ≈ 44 m^3/min

Cooling load = 4.69 tons

In interview room 2 loads is less than 5 tons. So there is no need to design the duct. Here FCU is installed.

15. Seminar room

By pass factor = 0.12

Room temperature = 23°C

ADP = 9°C

RSH = 16495.36 W

Room Rise = (1 – 0.12)*(296 - 282)

= 12.32

Dehumidified Air = 16495.36 / (20.44 * 12.32)

= 65.50 m^3/min

Safety factor (5%) = 3.27 m^3/min

Total dehumidified air = 68.77 m^3/min ≈ 69 m^3/min

Cooling load = 7.16 tons

Initial velocity = 300 m/min

Duct area = $\frac{69}{300}$ = 0.23 m^2 = 2.47 ft^2

Duct size = 24 * 16 inch = 600 * 400 mm

Equivalent duct diameter = 21.3 inch = 532.5 mm ≈ 530 mm

Friction drop = 0.0601

16. Central design office

By pass factor = 0.12

Room temperature = 23°C

ADP = 10°C

RSH = 55041.21 W

Room Rise = (1 – 0.12)*(296 - 283)

= 11.44

Dehumidified Air = 55041.21 / (20.44 * 11.44)

= 235.38 m^3/min

Safety factor (5%) = 11.76 m^3/min

Total dehumidified air = 247.14 m^3/min ≈ 247 m^3/min

Cooling load = 24.69 tons

Initial velocity = 300 m/min

Duct area = $\frac{247}{300}$ = 0.82 m^2 = 8.82 ft^2

Duct size = 38 * 36 inch = 950 * 900 mm

Equivalent duct diameter = 40.4 inch = 1010 mm ≈ 1000 mm

Friction drop = 0.0236

17. Auditorium

By pass factor = 0.12

Room temperature = 23°C

ADP = 9°C

RSH = 101211.71 W

Room Rise = (1 – 0.12)*(296 – 282)

= 12.32

Dehumidified Air = 101211.71 / (20.44 * 12.32)

= 401.91 m^3/min

Safety factor (5%) = 20.09 m^3/min

Total dehumidified air = 422 m^3/min

Cooling load = 50.14 tons

Initial velocity = 300 m/min

Duct area = $\frac{422}{300}$ = 1.40 m^2 = 15.06 ft^2

Duct size = 48 * 48 inch = 1200 * 1200 mm

Equivalent duct diameter = 52.6 inch = 1315 mm ≈ 1300 mm

Friction drop = 0.0318

18. Library Facility

By pass factor = 0.12

Room temperature = 23°C

ADP = 9°C

RSH = 11554.81 W

Room Rise = (1 – 0.12)*(296 – 282)

= 12.32

Dehumidified Air = 11554.81 / (20.44 * 12.32)

= 45.88 m^3/min

Safety factor (5%) = 2.29 m^3/min

Total dehumidified air = 48.17 m^3/min ≈ 48 m^3/min

Cooling load = 5.57 tons

Initial velocity = 300 m/min

Duct area = $\frac{48}{300}$ = 0.16 m^2 = 1.72 ft^2

Duct size = 18 * 16 inch = 450 * 400 mm

Equivalent duct diameter = 18.5 inch = 462.5 mm ≈ 460 mm

Friction drop = 0.0318

CHAPTER – 5

RESULT ANALYSIS

The result analysis is based on the duct design of the TIIR building with hand calculation and duct design software like ductulator.

5.1. Duct size:

To design the duct for TIIR building calculation of cooling load and air flow rate is done. By taking some suitable velocity (from Table 3.1), considering noise factor main duct area is calculated. Based on these duct area, the duct size is find out (from Fig. 3.3) for the rectangular duct as well as round duct. The cooling load, dehumidified air flow, duct size for all room is given in below:

Table 5.1 – Cooling load and dehumidified air for respective room

S.N.	Room Name	Cooling Load (tons)	Dehumidified Air Flow (m^3/min)	Type of unit used (FCU/AHU)
1	120 seat Lecture Room 1	12.39	113	AHU
2	Direct TIIR	2.11	21	FCU
3	Admin Office	3.03	32	FCU
4	Placement Office	2.27	25	FCU
5	IPR Office	2.93	32	FCU
6	120 seat Lecture Room 2	12.39	113	AHU
7	Office Room	6.96	69	AHU
8	Meeting Room	6.77	66	AHU
9	Library	4.53	43	FCU
10	Dining	5.85	63	AHU
11	Alumini Relation	4.68	49	FCU
12	Alumini Visitor	7.55	74	AHU
13	Interview Room 1	2.95	31	FCU
14	Interview Room 2	4.69	44	FCU
15	Seminar Room	7.16	69	AHU
16	Central Design Office	24.69	247	AHU
17	Auditorium	50.14	422	AHU
18	Library Facility	5.57	48	AHU

Table 5.2 – duct size comparison between hand calculation and ductulator software

S.N.	Room	Hand calculation			Using software (ductulator)		
		Rectangular duct (mm)	Round duct (mm)	Friction drop	Rectangular duct (mm)	Round duct (mm)	Friction drop
1	120 seat lecture room 1	650 * 600	680	0.0445	700 * 550	675	0.0487
2	Direct TIIR	FCU is used, no ducting is required					
3	Admin Office	FCU is used, no ducting is required					
4	Placement office	FCU is used, no ducting is required					
5	IPR Office	FCU is used, no ducting is required					
6	120 seat lecture room 2	650 * 600	680	0.0445	700 * 550	675	0.0487
7	Office Room	600 * 400	530	0.0600	550 * 450	525	0.0655
8	Meeting Room	500 * 450	520	0.0630	500 * 450	520	0.0674
9	Library	FCU is used, no ducting is required					
10	Dining	550 * 400	510	0.0663	500 * 450	500	0.0694
11	Alumini Relation	FCU is used, no ducting is required					
12	Alumini Visitor	550 * 500	570	0.0603	550 * 450	550	0.0627
13	Interview Room 1	FCU is used, no ducting is required					
14	Interview Room 2	FCU is used, no ducting is required					
15	Seminar Room	600 * 400	530	0.0623	550 * 450	530	0.0655
16	Central Design Office	950 * 900	1000	0.0295	1000 * 850	1000	0.0303
17	Auditorium	1200*1200	1300	0.0195	1250*1150	1300	0.0220
18	Library Facility	450 * 400	460	0.0792	450 * 400	460	0.0822

2. For calculating duct size equal friction method is used. Frictional pressure drop are different for all rooms (given in above table) as velocity kept constant.

3. Ansys 13.0 is used to observe the friction loss in rectangular duct as well as circular duct. For analysis we select only small portion of duct (3 m), also it can be applied for all ducting in a building.

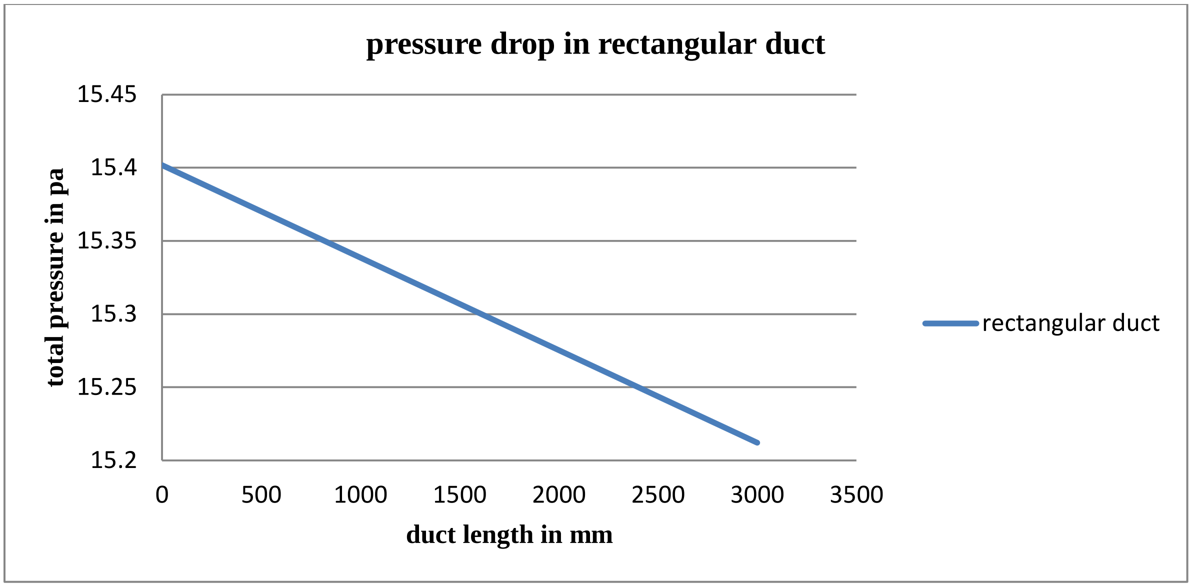

Fig. 5.1 – Pressure loss due to rectangular duct

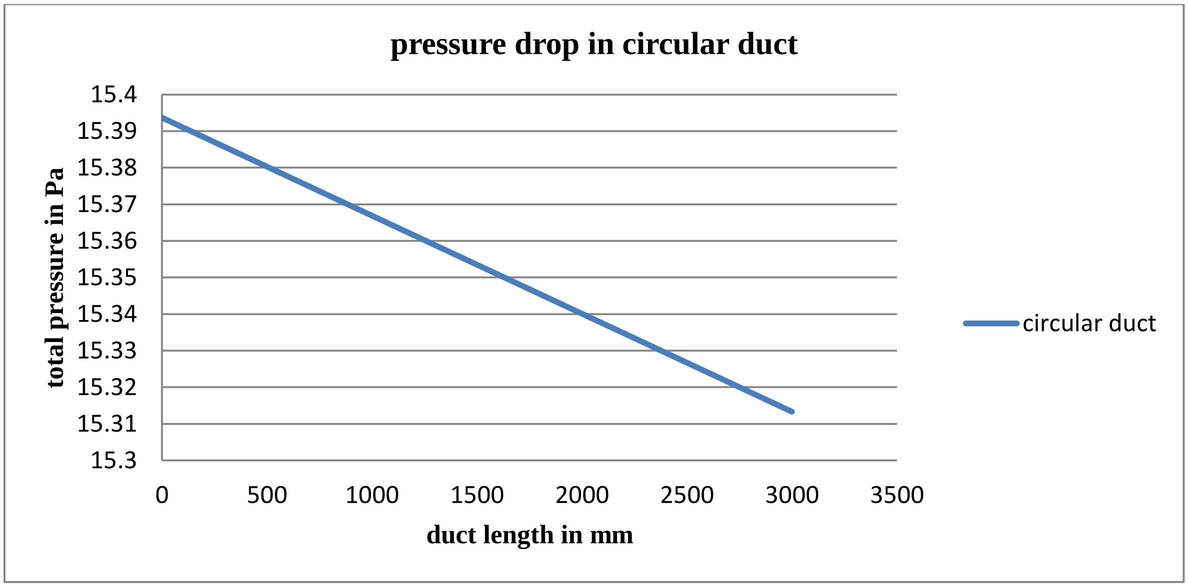

Fig. 5.2 – Pressure loss due to circular duct

After studying Fig. 5.1 and fig. 5.2 the result comes that circular duct has minimum friction loss as compared to the rectangular duct.

CHAPTER – 6

CONCLUSION

The following conclusion summarizes the design work presented in this thesis:-

1. The duct design for TIIR building is done, by using equal friction method. All values are comparable with duct software called ductulator.
2. The calculated value of frictional is less or near as calculated by software. Due to less value of friction drop, duct diameter is increased but loss in total pressure (i.e. static pressure, velocity pressure) can be avoided.
3. Due to increased duct diameter the use of damper may be decreased.
4. Also the circular duct can carry more air in less space, because of that, less duct material, less duct surface friction and less insulation is required.
5. Pressure loss in duct fitting can be minimized by proper design the elbow shape.
6. Ansys 13.0 software is used to analyze the pressure loss in circular and rectangular duct. After analysis we conclude that the circular duct has minimum friction loss, so it is better shape for ducting.

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